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EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER CHARACTERISTICS OF THE HEAT PIPE

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Original Article

ABSTRACT

The rapid development of electronic industries and increasing processing speed of components, decrease the heat transfer surface area. This generates terribly high heat fluxes, leading to massive temperature rises. To remove the heat flux from the components, heat pipes are being used, now a days. The reliability of electronic components decreases by 10% , for every increasing 2°C raise, above normal operating temperature. Therefore, it's necessary to develop a replacement reasonably sink, to satisfy those necessities, which ought to have the next heat transfer performance, lighter weight and smaller volume. Heat pipe may be a promising candidate, to satisfy the preceding thermal management challenges, within the future.

In this paper, two heat pipes were studied through experiment exploitation deionised water, because of the operating fluid. The wick used in the heat pipes is made of SS304 material, with 100 mesh and 200 meshes, respectively. The heat pipes were positioned at totally different angles 0^0, 45^0, 90^0 with horizontal. Experiments were conducted with the natural convection, forced convection condenser cooling, with different heating fluxes. The thermal resistances of heat pipe, operating with forced convection cooling, with 200 mesh at 90^o inclination is 53%, lower than the heat pipe operating at horizontal at 50W heat input. The heat transfer coefficient of heat pipe, operating at 45^o angle is 71% higher than heat pipe operating at horizontal position, with forced convection cooling with 200 meshes. The effective thermal conductivity of heat pipe, operating at 45^o is 47% higher, the heat pipe operating with forced convection cooling, with 200 mesh at horizontal position with 50W heat input.

KEYWORDS: Heat Pipe, Forced Convection, Mesh & Copper – Water Heat Pipe

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INTRODUCTION

Heat pipe performance is dependent relative on its geometry, the working fluid, the operating temperature, the capillary wicking material, and the applied heat flux. The three basic factors that determine the liquid driving potential, and thereby the heat transport capability are the surface tension, the contact angle and the geometry of the solid surfaces, at the three phase (solid-liquid-Vapour) boundary line. The higher the surface tension and lower the solid-liquid contact angle, the higher the driving potential and heat transport capability [1].

The dry out phenomenon can be found out, by increasing the evaporator temperature drastically [7, 8]. The capillary or wicking restriction that happens once the pumping rate of the wicking material isn't enough to produce. Disadvantage of the warmth pipes area unit, that the effective space is circular, so the warmth dissipation

over rectangular heat sinks, can decrease towards their corners as the thermal resistance will have no two –phase component, in these regions. [3, 6, 10].

METHODOLOGY

The heat pipes were constructed using copper tubing, with an outer diameter of 12.7mm, and inner diameter 9.52mm, and the heat pipes were 450 mm long, and the one end was closed with 3.2 mm thick copper end cap, that were soldered with lead tin solder. Another end of the heat pipe was connected to the tube, which the pressure gauges, vacuum pump connected. The wicks for each heat pipes, were made from a woven stainless steel wire screen mesh, with 100 mesh and 200 mesh each, had two layers of screen mesh wicks. Tests were performed here, for heat pipes as shown in figure 1 that included fluid loading corresponding to 50% of the amount of inner volume of the evaporator. The heat was applied to the heat pipes, for 100 mm length, which is the evaporator section connected to the dimmer stat, the adiabatic section is covered with zirconia sheet. T-type thermocouples were inserted in their locations, by the thermo wells. The thermometers were covered with eraldide, at the outer end of the thermo well. Thermocouples were inserted into the wall, to measure the axial vapour temperature. The uncertainty of the thermocouples was $\pm 0.1^{O}$C.

Figure 1: Heat Pipe Experimental Setup

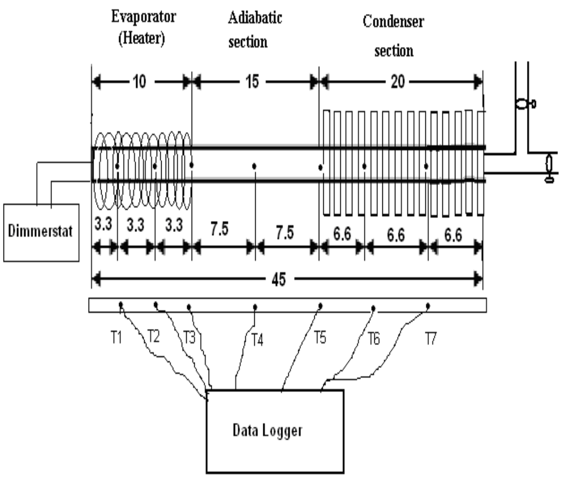

Figure 2: Line Diagram of Heat Pipe Experimental Setup

The heater power was then enlarged, and measurements of temperature were taken once the warmth pipe had reached steady state. The transfer of heat rate into the evaporator was calculable, by subtracting estimates of natural

convection losses, from the heater and from the measured wattage of the heater. The natural convection losses round the insulated evaporator section, were calculable discrimination.

The correlation was located to be, but third of the measured wattage in the least power settings. The output heat transfer rate from the condenser was computed, by applying an energy balance to the condenser flow. The effects of heat losses to the ambient and viscous heating effects, were characterized independently, and were subtracted from the output heat transfer rate. The three way valve, allowed for the removal of non condensable gases from the wick test, stand prior to filling, via a vacuum pump. The working fluid used was lab grade deionised water. It is employed because, the operating fluid, as water incorporates a comparatively massive physical phenomenon.

RESULTS AND DISCUSSIONS

Figure 3, shows the variation of thermal resistance, with heat input at different tilt angles of the forced convection heat pipe, with 200 mesh size. At 50W of heat input, thermal resistances are 0.2726(90), 0.168(45),and 0.584(horizontal).The thermal resistance was least for heat pipe, operating at 45 degrees. From the test results, are shown in figure and it is seen that, the thermal resistance of the heat pipe decreases, as the thermal load increases. The thermal resistances of heat pipe, operating at 90^{o} inclination is 53%, lower than the heat pipe operating at horizontal, at 50W heat input.

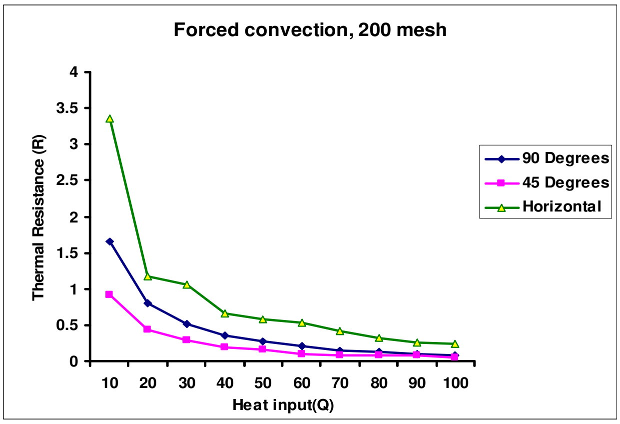

Figure 3: Variation of Thermal Resistance of Heat Pipe with Heat Input

Figure 4, shows the variation of heat transfer coefficient with heat input, at different tilt angles of the forced convection heat pipe, with 200 mesh size. At 50W of heat input, heat transfer coefficients are 921.7(90), 1488.4(45), 430.28 (horizontal). The transfer coefficient for the heat pipe, operating at 45 degrees dissipates more amount of heat. The heat transfer coefficient of heat pipe operating at 45^{o} is 71% higher than heat pipe, operating at horizontal position.

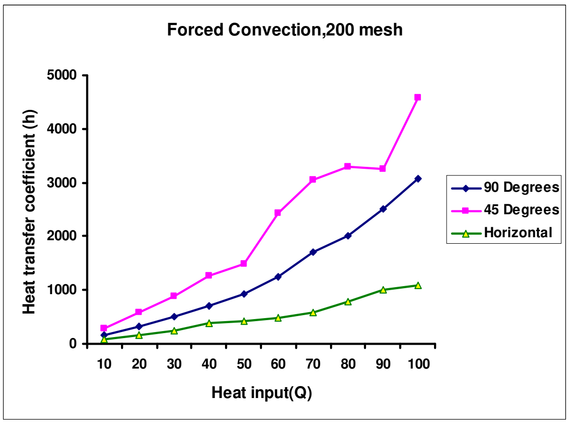

Figure 4: Variation of Heat Transfer Coefficient with Heat Input

Figure 5, shows the variation of vapour temperature, with respect to the time. It can be seen, it is about 30 minutes for the test system to reach steady state, when power was 50W. All the readings are taken and the system had reached its steady state. Initially, the reading was 23.6^{o}C, after vacuuming the tube.

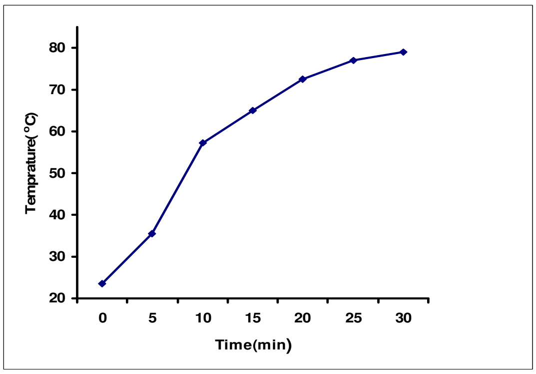

Figure 5: Variation of Vapour Temperature at Starting of Heat Pipe

Figure 6, shows the variation of temperature difference with heat input. ΔT increases with increase in the heat load, at the same condenser temperature. It increases with condenser temperate, for the same heat load. The maximum temperature difference of heat pipe, operating at horizontal is 72% higher, and the heat pipe operating at 45^{o} at heat input 50W.

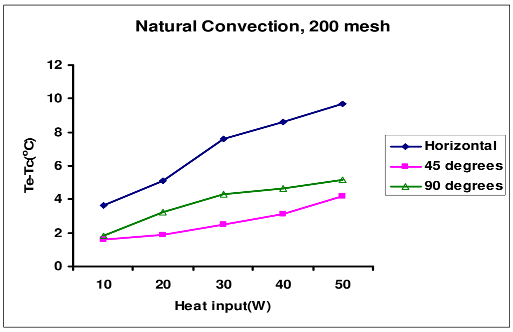

Figure 6: Variation of Temperature Difference with Heat Input

Figure 7, shows the variation of thermal conductivity of the heat pipe, with heat input, the effective thermal physical phenomenon of the pipe and the performance of the load of heat, for different condenser temperatures

(natural, forced convection), and inclination angles. The maximum value of thermal conductivity at 45 degrees inclination angle is 405.5W/m°C. The effective thermal conductivity of heat pipe, operating at 45° is 47% higher, than the heat pipe operating at horizontal.

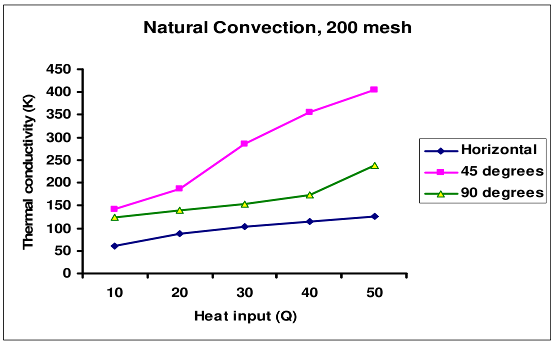

Figure 7: Variation of Thermal Conductivity of Heat Pipe with Heat Input

Figure 8, shows the variation of vapour temperature, along the length of pipe. The vapour temperatures are the maximum for natural convection heat pipe. The vapour temperatures of heat pipe, operating with forced convection with 200 mesh is 42%, lower than heat pipe operating at natural convection 200 mesh at adiabatic section**.**

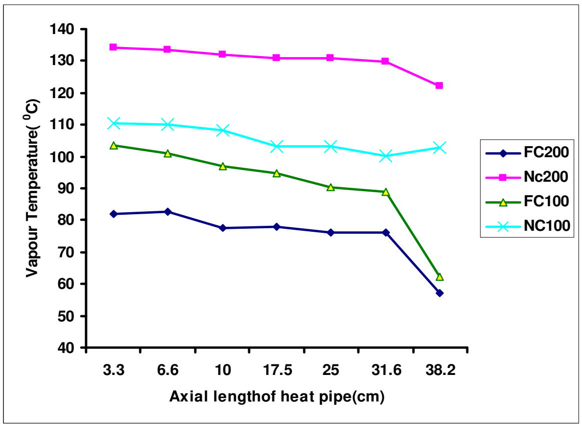

Figure 8: Vapour Temperature along the Length of Heat Pipe

Equations

The effective heat pipe thermal resistance, at a given design power from equation 1 is as follows:

$$R = \frac{(Te - Tc)}{Q} \quad (1)$$

Thermal conductivity of the heat pipe system is determined, by the following equation 2 as follows:

$$K = LQ/(\Delta T.A) \quad (2)$$

CONCLUSIONS

This paper discusses the thermal improvement of pipe performance exploitation, using water as working fluid. In this case, deionised water was used as working fluid. The results of the performance test are as follows

- The heat transfer coefficient increases and thermal resistance of the heat pipe decreases, for the forced convection

compared to the natural convection. The thermal resistance of heat pipe operating at 90o inclinationm is 53% lower than the heat pipe, operating at horizontal at 50W heat input.

- The overall heat transfer coefficient was higher, for the heat pipe with forced convection than natural convection. The heat transfer coefficient of heat pipe, operating at 45^o is 71% higher than heat pipe operating at horizontal position. The transfer coefficient for the heat pipe, operating at 45 degrees dissipates more amount of heat.
- The effective thermal conductivity of the heat pipe, increases as the heat input increases. The effective thermal conductivity of heat pipe, operating at 45^o is 47%, higher than the heat pipe operating at horizontal.

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